(19) Europäisches Patentamt
European Patent Office
Office européen des brevets

(11) EP 0 844 385 A1

(12) **EUROPEAN PATENT APPLICATION**

(43) Date of publication:
27.05.1998 Bulletin 1998/22

(51) Int. Cl.[6]: **F02M 47/02**

(21) Application number: **97120502.6**

(22) Date of filing: **21.11.1997**

(84) Designated Contracting States:
AT BE CH DE DK ES FI FR GB GR IE IT LI LU MC NL PT SE
Designated Extension States:
AL LT LV MK RO SI

(30) Priority: **21.11.1996 JP 310470/96**
25.11.1996 JP 313328/96

(71) Applicant: **DENSO CORPORATION**
Kariya-City Aichi-Pref. 448 (JP)

(72) Inventors:
- **Murakami, Masashi**
  Kariya-city, Aichi-pref. 448 (JP)
- **Toyao, Tetsuya**
  Kariya-city, Aichi-pref. 448 (JP)
- **Matsumoto, Shuichi**
  Kariya-city, Aichi-pref. 448 (JP)

(74) Representative:
KUHNEN, WACKER & PARTNER
Alois-Steinecker-Strasse 22
85354 Freising (DE)

(54) Accumulator fuel injection apparatus for internal combustion engine

(57) An accumulator fuel injection apparatus for an internal combustion engine is provided which includes a solenoid-operated fuel injector (1). The fuel injector (1) includes a solenoid valve (20) and a needle valve (220). The solenoid valve (20) establishes and blocks fluid communication between a pressure control chamber (30) supplied with fuel pressure from a fuel inlet (70) and a drain passage (69) formed in a valve body (91) to change fuel pressure within the pressure control chamber (30), thereby bringing the needle valve (220) into engagement with and disengagement from a spray hole (101a). The fuel injector (1) also has a first orifice disc (210) and a second orifice disc (211) installed within the valve body (91). The first orifice disc (210) has formed therein a first orifice (66) which provides a first flow resistance to fuel flowing from the fuel inlet (70) into the pressure control chamber (30). Similarly, the second orifice disc (211) has formed therein a second orifice (67) which provides a second flow resistance smaller than the first flow resistance to the fuel flowing out of the pressure control chamber (30) into the drain passage (69). The second orifice (211) disc is disposed on the first orifice disc (210) so that thicknesswise directions thereof coincide with each other.

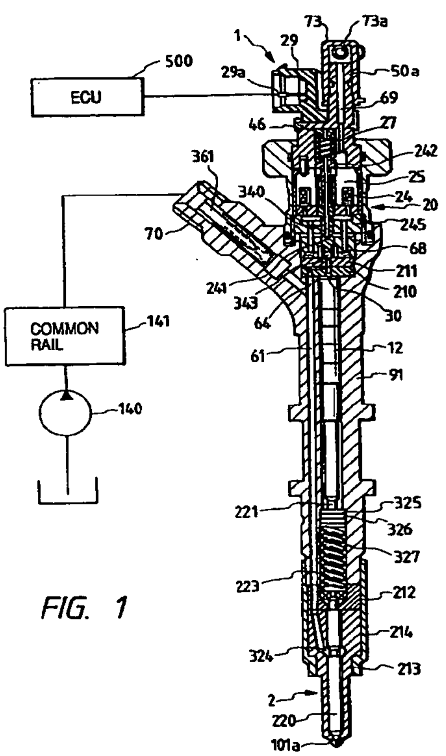

FIG. 1

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Description

BACKGROUND OF THE INVENTION

The present invention relates generally to an accumulator fuel injection apparatus equipped with a solenoid valve for injecting fuel stored within a common rail (i.e., surge tank) at a high pressure level into an internal combustion engine.

U.S.P. No. 4,798,186 to Ganser, issued on January 17, 1989 and U.S.P. No. 5,660,368 to De Matthaeis et al., issued on August 26, 1997 disclose electromagnetically controlled fuel injection systems designed to accumulate the fuel within a common rail under pressure through a high-pressure feed pump and inject the fuel into an internal combustion engine. These fuel injection systems use a fuel injector and a solenoid operated two-way valve. The fuel injector includes a pressure control chamber communicating with a high-pressure fuel passage. The two-way valve selectively establishes and blocks fluid communication between the pressure control chamber and a low-pressure chamber to control the fuel pressure acting on a needle valve of the fuel injector for opening and closing a spray hole.

Between the high-pressure fuel passage and the pressure control chamber, a first orifice is formed in a first orifice member to restrict the flow rate of fuel entering the pressure control chamber from the high-pressure fuel passage. A second orifice is also formed in a second orifice member between the pressure control chamber and the low-pressure chamber to restrict the flow rate of fuel flowing from the pressure control chamber to the low-pressure chamber when the solenoid operated two-way valve is opened. When a response rate of the solenoid operated two-way valve is not changed at valve closing and opening, fuel injection characteristics such as injection timing, injection quantity, and rate of injection almost depend upon the flow rate characteristics of the first and second orifices.

Of the fuel injection characteristics, the quantity of fuel at the injection beginning, at the injection end, and during an early part of injection is determined by a difference in flow rate of fuels flowing from the high-pressure fuel passage to the pressure control chamber and flowing from the pressure control chamber to the low-pressure chamber when the solenoid operated two-way valve is opened. The quantity of fuel flowing out of the fuel injector after termination of injection and an interval between a time when the rate of injection shows a peak value and termination of injection (hereinafter, referred to as an injection cut-off period) are determined by the flow rate of fuel flowing from the high-pressure fuel passage to the pressure control chamber after the solenoid operated two-way valve is turned off or closed. Therefore, in order to ensure desired injection characteristics, it is necessary to adjust the flow rate characteristics of the first and second orifices by replacing the first and second orifice plates.

Since the fuel injection characteristics such as the injection timing, the injection quantity, and the rate of injection are, as described above, almost determined based on the flow rate characteristics of the first and second orifices, they will be changed greatly depending upon the shape, sectional area, circularity, inlet dimension, outlet dimension, surface roughness of the first and second orifices.

The optimum fuel injection over a wide range of engine operation which limits the rate of injection at an early part of injection and stops the injection at a high response rate, requires finely drilling the first and second orifices to have a diameter of approximately Ø0.2 mm to Ø0.4 mm.

In the De Matthaeis et al. system (U.S.P. No. 5,660,368), the first and second orifices are formed in a single injector component. Thus, both the first and second orifices must be replaced even when it is required to change the flow rate characteristics of either of the first and second orifices for adjusting the injection timing and/or the injection characteristics at early and/or late part of injection. This leads to the problem that production yield of injector components for injection characteristic adjustment is decreased. Further, variations in machining accuracy in forming the first and second orifices may mutually affect, thereby making it more difficult to ensure the desired injection characteristics. This also increases the number of times the injector component is replaced until the desired injection characteristics are obtained in an injection characteristics adjustment process.

In the Ganser's system (U.S.P. No. 4,798,186), the first and second orifices are formed in different injector components and thus may be replaced separately for changing the flow rate characteristics. One of the injector components having formed therein either of the first and second orifices supports the other slidably. A clearance between sliding surfaces of the injector component pair having formed therein the first and second orifices is decreased as much as possible to facilitate sealing thereof for avoiding leakage of the high-pressure fuel out of the pressure control chamber. Therefore, replacement of only one of the injector component pair may result in an undesirable decrease in the clearance, thereby precluding the sliding motion of the injector components or in great increase in the clearance, thereby leading to the leakage of fuel.

SUMMARY OF THE INVENTION

It is therefore a principal object of the present invention to avoid the disadvantages of the prior art.

It is another object of the present invention to provide an improved structure of a fuel injector apparatus for an internal combustion engine which is designed to obtain desired injection characteristics in a simple and economical manner.

According to one aspect of the present invention,

there is provided an accumulator fuel injection apparatus for injecting high-pressure fuel stored within a common rail into an internal combustion engine which comprises: (a) a valve body having formed therein a fuel inlet passage and a spray hole, fuel inlet passage communicating with the common rail; (b) a valve member disposed slidably within the valve body for selectively establishing and blocking fluid communication between the fuel inlet passage and the spray hole; (c) a pressure control chamber formed within the valve body, the pressure control chamber being connected to the fuel inlet passage to introduce therein fuel pressure which acts on the valve member to block the fluid communication between the fluid inlet passage and the spray hole; (d) a fuel pressure drain passage formed within the valve body, connected to the pressure control chamber for draining the fuel pressure out of the valve body; (e) a solenoid valve selectively establishing and blocking fluid communication between the pressure control chamber and the fuel pressure drain passage; (f) a first orifice plate having formed therein a first orifice which provides a first flow resistance to fuel flowing from the fuel inlet passage into the pressure control chamber; and (g) a second orifice plate having formed therein a second orifice which provides a second flow resistance smaller than the first flow resistance to the fuel flowing out of the pressure control chamber into the fuel pressure drain passage when the solenoid valve establishes the fluid communication between the pressure control chamber and the fuel pressure drain passage, the second orifice plate being disposed on the first orifice plate so that thicknesswise directions thereof coincide with each other.

In the preferred mode of the invention, the first orifice has a length extending in parallel to a thickness of the first orifice plate. The second orifice has a length extending in parallel to a thickness of the second orifice plate.

The first and second orifices are formed by drilling the first and second orifice plates and reaming drilled holes.

The first and second orifices may alternatively be machined in an electron discharge method.

The first and second orifices may also be polished by forcing an abrasive solution made of a mixture of liquid and abrasive grain therethrough until the flow of the abrasive solution through the first and second orifices reaches a given flow rate.

Each of the first and second orifice plates Is made of a disc in which first and second through holes are formed. Two knock pins are inserted into the valve body through the first and second through holes of the first and second orifice plates to fix angular positions of the first and second orifice plates relative to the valve body.

The first, and second through holes are formed at different intervals away from the center of each of the first and second orifice plates so that a line extending through the centers of the first and second through holes is offset from the center of each of the first and second orifice plates.

A first large-diameter hole which has a diameter greater than that of the first orifice may be formed in the first orifice plate coaxially with the first orifice in communication with the first orifice. A second large-diameter hole which has a diameter greater than that of the second orifice may also be formed in the second orifice plate coaxially with the first orifice in communication with the second orifice.

The first and second orifice plates are so disposed within the valve body that the first orifice plate is exposed at a first surface to the pressure control chamber and at a second surface opposite the first surface in contact with a first surface of the second orifice plate, and the second orifice plate is exposed at a second surface opposite the first surface to the fuel pressure drain passage. A cylindrical chamber is formed in the second surface of the second orifice plate in communication with the second orifice which has a diameter greater than that of the second orifice.

The solenoid valve includes a valve head which opens and closes the second orifice to establish and block the fluid communication between the pressure control chamber and the fuel pressure drain passage. An annular valve seat on which the valve head of the solenoid valve is to be seated to block the fluid communication between the pressure control chamber and the fuel pressure drain passage, is formed on the second surface of the second orifice plate around an opening of the cylindrical chamber.

An annular groove is formed in the second surface of the second orifice plate around the annular valve seat of the second orifice plate in fluid communication with the fuel pressure drain passage.

The cylindrical chamber may alternatively be formed in the valve head opening to the second orifice of the second orifice plate which has a diameter greater than that of the second orifice.

BRIEF DESCRIPTION OF THE DRAWINGS

The present invention will be understood more fully from the detailed description given hereinbelow and from the accompanying drawings of the preferred embodiment of the invention, which, however, should not be taken to limit the invention to the specific embodiment but are for explanation and understanding only.

In the drawings:

Fig. 1 is a cross sectional view taken along the line I-I in Fig. 4 which shows a fuel injector incorporated in a fuel injection apparatus for an internal combustion engine according to the first embodiment of the Invention;

Fig. 2 is a partial cross sectional view which shows a major portion of the fuel injection in Fig. 1;

Fig. 3 is a cross sectional view taken along the line

III-III in Fig. 4;

Fig. 4 is a plan view which shows a fuel injector according to the first embodiment of the invention;

Fig. 5(a) is a plan view which shows a first orifice plate mounted in the fuel injector in Fig. 1;

Fig. 5(b) is a cross sectional view taken along the line B-B in Fig. 5(a);

Fig. 6(a) is a plan view which shows a second orifice plate mounted in the fuel injector in Fig. 1;

Fig. 6(b) is a cross sectional view taken along the line B-B in Fig. 6(a);

Fig. 7(a) is a time chart which shows a displacement of a movable member of a solenoid valve incorporated within the fuel injector in Fig. 1;

Fig. 7(b) is a time chart which shows a variation in pressure within a pressure control chamber formed in the fuel injector in Fig. 1;

Fig. 7(c) is a time chart which shows a displacement of a control piston mounted in the fuel injector in Fig. 1;

Fig. 7(d) is a time chart which shows a variation in rate of injection;

Fig. 8 is a partial cross sectional view which shows a major portion of a fuel injector according to the second embodiment of the invention;

Fig. 9 is a plan view which shows a second orifice plate of a fuel injector according to the third embodiment of the invention;

Fig. 10 is a plan view which shows a first orifice plate of a fuel injector according to the third embodiment of the invention;

Fig. 11(a) is a cross sectional view which shows an end of a valve shaft of a solenoid valve of a fuel injector according to the fourth embodiment of the invention;

Fig. 11(b) is a partial perspective view which shows first and second orifice plates of a fuel injector according to the fourth embodiment of the Invention;

Fig. 12(a) is a cross sectional view which shows an end of a valve shaft of a solenoid valve of a fuel injector according to the fifth embodiment of the invention; and

Fig. 12(b) is a partial perspective view which shows first and second orifice plates of a fuel injector according to the fifth embodiment of the invention.

DESCRIPTION OF THE PREFERRED EMBODIMENT

Referring now to the drawings, particularly to Fig. 1, there is shown a fuel injection apparatus for a diesel engine equipped with a solenoid-operated fuel injector 1 according to the first embodiment of the invention.

The fuel injector 1 is connected at an inlet port 70 to a common rail 141 through a fuel supply pipe. To the common rail 141, high-pressure fuel is supplied through a fuel pump 140. A control signal is inputted to a pin 29a of a wire harness connector 29 from an electronic control unit (ECU) 500 for controlling the fuel injection into a combustion chamber of the engine.

The fuel injector 1 includes a spray nozzle 2 and an injector body 91. The spray nozzle 2 includes a nozzle body 213 having a spray hole 101a formed in the tip thereof. A needle valve 220 is slidably disposed within the nozzle body 213 to close and open the spray hole 101a. The nozzle body 213 and the injector body 91 are jointed through a packing chip 212 by a retaining nut 214. A pressure pin 221 and a control piston 12 are disposed within the injector body 91 in alignment with the needle valve 220. The control piston 12 is in contact with the pressure pin 221, but may alternatively be bonded thereto. The pressure pin 221 is disposed within a spring 223. The spring 223 urges the pressure pin 221 downward, as viewed in the drawing, to bring the needle valve 220 into constant engagement with the spray hole 101a. The set load of the spring 223 is adjusted by load adjusting spacers 325 and 326. The control piston 22 is exposed at an end opposite to the spray hole 101a to a pressure control chamber 30.

The high-pressure fuel entering the inlet port 70 passes through a fuel filter 361 and flows both to high-pressure fuel passages 61 and 64. The part of the high-pressure fuel entering the high-pressure passage 61 is supplied directly to an annular fuel sump 324 formed around the periphery of the needle valve 220, while the other entering the high-pressure fuel passage 64 is supplied to the pressure control chamber 30. The pressure of fuel in the fuel sump 324 acts on the needle valve 220 to lift it upward, as viewed in the drawing, for establishing fluid communication between the fuel sump 324 and the spray hole 101a, while the pressure of fuel in the pressure control chamber 30 acts on the control piston 12 to urge the needle valve 220 downward so that it closes the spray hole 101a.

The injector body 13 has also formed therein a fuel drain passage 365, as clearly shown in Fig. 3, which communicates with a spring chamber 327 and drains the fuel leaking out of sliding clearances between inner walls of the injector body 91 and the spray nozzle 2 and outer peripheral surfaces of the control piston 12 and the needle valve 220 to a low-pressure fuel chamber 68 through fuel passages 210b and 211b, as clearly shown in Fig. 2, formed in first and second orifice plates 210 and 211, as will be described later in detail. The fuel within the low-pressure fuel chamber 68 passes through low-pressure fuel passages 345a formed in a valve cylinder 345, a low-pressure fuel passage 341a formed in a valve shaft 241, a low-pressure fuel passage 242a formed in a plunger 242, holes 334a formed in an armature 26 of a solenoid valve 20, a low-pressure fuel passage 25a extending along the center of a core 25 of the solenoid valve 20, and a low-pressure fuel passage 69 formed in a housing 50 and then flows out of a fuel withdrawal union 73 through a low-pressure fuel passage 73a, as shown in Fig. 1 so that excess fuel is drained outside the fuel injector 1.

The first and second orifice plates 210 and 211 are, as clearly shown in Fig. 2, disposed adjacent each other so that thicknesswise directions thereof coincide with each other and retained by the valve cylinder 345 within the injector body 91. The first orifice plate 210 has formed therein a first orifice 66 which restricts the flow rate of fuel from the high-pressure fuel passage 64 to the pressure control chamber 30. The second orifice plate 211 has a second orifice 67 formed in the center thereof which limits the flow rate of fuel from the pressure control chamber 30 to the low-pressure fuel chamber 68. The first and second orifice plates 210 and 211 are, as shown in Figs. 5(a) to 6(b), made of discs. The first and second orifices 66 and 67 communicate with large-diameter holes 66a and 67a formed in bottoms of the first and second orifice plates 210 and 211 coaxially with the first and second orifices 66 and 67 and extend parallel to vertical center lines (i.e., the thicknesswise directions) of the first and second orifice plates 210 and 211, respectively, so that they are easy to machine with high accuracy.

The first orifice plate 210 has formed therein two bores 210a. Similarly, the second orifice plate 211 has formed therein two bores 211a. The bores 210a are arranged at the same interval away from the center of the first orifice plate 210 so that a line extending through the centers of the bores 210a is offset from the center of the first orifice plate 210. Similarly, the bores 211a are arranged at the same interval away from the vertical center line of the second orifice plate 211 so that a line extending through the centers of the bores 211a is offset from the center of the second orifice plate 211. Two positioning knock pins 55 (only one is shown in Fig. 3 for the brevity of illustration) are inserted into the injector body 91 through the bores 210a and 211a of the first and second orifice plates 210 and 211 which are aligned with each other. This fixes the positional relation between the first and second orifice plates 210 and 211 and the injector body 91 and also brings the fuel passages 210b and 211b formed in the first and second orifice plates 210 and 211 into coincidence with each other. The valve cylinder 345 and the injector body 91 are connected in screw fashion.

The second orifice plate 211 has, as shown in Figs. 2 and 6(b), an annular flat surface 211c formed on an upper surface around the center thereof (i.e., the second orifice 67). The annular flat surface 211c works as a valve seat on which a ball 243 (i.e., a valve head), as will be described later in detail, of the solenoid valve 20 is seated. When the ball 243 is seated on the annular flat surface 211c, it blocks the fluid communication between the pressure control chamber 30 and the low-pressure fuel chamber 68. An annular path 155 is formed around the annular flat surface 211c which adds a given volume to the low-pressure fuel chamber 68 for facilitating ease of the fuel flow to the low-pressure fuel chamber 68 when the ball 243 is lifted away from the second orifice plate 211.

The first and second orifices 66 and 67 may be formed by drilling the first and second orifice plates 210 and 211 and reaming the drilled holes or by drilling the first and second orifice plates 210 and 211 in the electrical discharge machining. The first and second orifices thus formed may also be polished in a finishing process by forcing an abrasive solution made of a mixture of liquid and abrasive grain therethrough until the flow of the abrasive solution through the first and second orifices 66 and 67 reaches a given flow rate.

The solenoid valve 20 is a two-way valve designed to selectively establish and block the fluid communication between the pressure control chamber 30 and the low-pressure fuel chamber 68. The solenoid valve 20 is, as shown in Figs. 1 and 2, installed in the injector body 91 by the retaining nut 59. A pin 153 is inserted into the housing 50 and the core 25 to fix an angular relation therebetween and also hold relative rotation of the core 25 and the housing 50 when the retaining nut 59 is fastened during assembly for preventing a rotational load from acting on feeder terminals 72 shown in Fig. 3.

The solenoid valve 20 includes, as shown in Fig. 2, a coil 24 and a movable member 240. The coil 24 is made of wire wound within an annular groove formed in the core 25 and supplied with the power through the pin or terminal 29a of the connector 29. The core 25 is formed with 0.2 mm-thick silicon steel plates laminated spirally and welded to a hollow cylinder 333 in which the plunger 242 is disposed. The movable member 240 includes the valve shaft 241, the plunger 242, the ball 243, and the support 244. The valve shaft 241 and the plunger 242 are urged into constant engagement with each other by the fuel pressure and spring pressure exerted from the pressure control chamber 30 and the spring 27, respectively, so that they are moved vertically together when the solenoid valve 20 is turned on and off. The plunger 242 is made of a non-magnetic stainless steel for eliminating a magnetic effect on a magnetic circuit. The valve shaft 241 is slidably supported within the valve cylinder 345 and is made from a wear resistant material such as a magnetic material because the valve shaft 241 is magnetically located out of the magnetic circuit. The armature 26 is mounted on an upper portion of the valve shaft 241 in a press fit at a given interval away from a lower end of the core 25 of the solenoid valve 20 and made from, for example, a silicon steel since it needs to work as part of the magnetic circuit rather than needing to have wear resistance and has formed therein a plurality of bores 334a for reducing the fluid resistance during movement. The armature 26 may alternatively be mounted on the valve shaft 241 in caulking, welding, or any other suitable manner.

The amount of lift of the movable member 240 may be adjusted by changing the thickness of a spacer 54. The movable member 240 is lifted upward until the valve shaft 241 reaches the lower end of the cylinder 333. The armature 26, when lifted up to the upper limit, faces the lower end of the core 25 through a given gap so that the

movable member 240 can be moved downward, as viewed in Fig. 2, quickly when the coil 24 is turned off.

The support 244 is made of a hollow cylindrical member and mounted on an end of the valve shaft 241 in a press fit or welding. The ball 243 is disposed rotatably within a chamber defined by an inner wall of the support 244 and a cone-shaped recess formed in the end of the valve shaft 241 with a clearance of several μm between itself and the inner wall of the support 244. The support 244 is caulked at an end thereof to retain the ball 243 therein. The ball 243 is made from ceramic or cemented carbide and has formed thereon a flat surface which is seated on the annular flat surface 211c, as shown in Fig. 6(b), of the second orifice plate 211 for closing the second orifice 67 to block the fluid communication between the pressure control chamber 30 and the low-pressure fuel chamber 68. The amount of lift of the valve shaft 241 is approximately 100 μm, which allows the ball 243 to face at the flat surface to the second orifice plate 211 at all times regardless of the vertical position of the valve shaft 241 and to be seated on the annular flat surface 211c to close the second orifice 67 completely even when the ball 243 and the second orifice 67 are somewhat shifted in relative angular position.

The plunger 242 is disposed slidably within the cylinder 33 with a clearance with the inner wall thereof which is greater than the above sliding clearance. The coil spring 27 is interposed between a spacer or shim 46 and a flange of the plunger 242 to urge the plunger 242 downward so that the ball 243 closes the second orifice 67. The spring pressure acting on the plunger 242 may be adjusted by changing the thickness of the shim 46.

This embodiment has the following specifications on major parts of the structure:

1. diameter of the first orifice 66 = ∅0.20 mm
2. diameter of the second orifice 67 = ∅0.32 mm
3. diameter of the control piston 12 = ∅5.0 mm
4. stroke of the movable member 240 = 0.10 mm
5. diameter of the needle valve 220 = ∅4.0 mm
6. seat diameter of the needle valve 220 (i.e., the diameter of a seat area of a head of the needle valve 220 exposed to the spray hole 101a) = ∅2.25 mm
7. set load of the spring 27 = 50 N
8. set load of the spring 223 = 40 N

In operation, when the coil 24 of the solenoid valve 20 is deenergized, the plunger 242 is forced downward, as viewed in Fig. 2, by the spring pressure of the coil spring 27. The ball 243 is seated on the second orifice plate 211 to block the fluid communication between the pressure control chamber 30 and the low-pressure fuel chamber 68.

The diameter of the second orifice 67 (corresponding to a seat diameter of the ball 243 when seated on the second orifice plate 211) is 0.32 mm, and the diameter **d**, as shown in Fig. 6(b), of a ball seat of the second orifice plate 211 on which the ball 243 is seated is 0.50 mm. Thus, if the fuel pressure supplied from the common rail 141 (= the pressure within the pressure control chamber 30) is 150 Mpa, then the fluid pressure urging the ball 243 in a valve-opening direction is 19.5 N which is smaller than the set load of the spring 47 urging the movable member 240 of the solenoid valve 20 in a valve-closing direction that is, as described above, 50 N, so that the movable member 240 is not lifted upward as long as the coil 24 is turned off.

Since the diameter of the control piston 12 is 5.0 mm, the diameter of the needle valve 220 is 4.0 mm, the seat diameter of the needle valve 220 is, as described above, 2.25 mm, a pressure-energized area of the control piston 12 is greater than that of the needle valve 220, and a difference therebetween is approximately 11 mm^2. Since the spring pressure of the coil spring 223 urges the needle valve 220 in the valve-closing direction, the sum of the fuel pressure within the pressure control chamber 30 urging the control piston 12 in the valve-closing direction and the spring pressure of the spring 223 is greater than the fuel pressure within the fuel sump 324 lifting the needle valve 220 upward as long as the coil 24 is turned off. Specifically, when the solenoid valve 20 is in an off-position, the needle valve 220 continues to close the spray hole 101a.

When the coil 24 of the solenoid valve 20 is energized, it produces an electromagnetic force of approximately 60 N attracting the armature 26, so that the sum of the electromagnetic force and the fuel pressure within the pressure control chamber 30 urging the movable member 240 in the valve-opening direction becomes greater than the spring pressure of the coil spring 27, thereby lifting the movable member 240 upward to move the ball 243 away from the second orifice plate 211. This establishes the fluid communication between the second orifice 67 and the low-pressure fuel chamber 68 so that the fuel within the pressure control chamber 30 flows into the low-pressure fuel chamber 68 through the second orifice 67. Since the flow resistance of the second orifice 67 is smaller than that of the first orifice 66, the fuel pressure within the pressure control chamber 30 drops immediately when the ball 243 is lifted up away from the second orifice 67. When the fuel pressure within the pressure control chamber 30 drops, and the sum of the fuel pressure within the pressure control chamber 30 urging the control piston 12 in the spray hole-closing direction and the spring pressure of the coil spring 223 becomes smaller than the fuel pressure within the fuel sump 324 lifting up the needle valve 220, it will cause the needle valve 220 to be moved away from the spray hole 101a to initiate fuel injection.

When a given injection end is reached, the coil 24 of the solenoid valve 20 is deenergized, so that the electromagnetic force attracting the armature 26 is decreased from 60 N to zero (0). This causes the mov-

able member 240 to be moved by the spring force of the spring 27 away from the coil 24 to bring the ball 243 into engagement with the second orifice 67. The fuel pressure within the pressure control chamber 30 is elevated by the fuel flowing from the high-pressure fuel passage 64 through the first orifice 66, so that the sum of the fuel pressure within the pressure control chamber 30 urging the control piston 12 in the spray hole-closing direction and the spring pressure of the spring 223 becomes greater than the fuel pressure within the fuel sump 324 lifting the needle valve 220 upward, thereby bringing the needle valve 220 into engagement with the spray hole 101a to terminate the fuel injection.

Figs. 7(a) to 7(d) show a displacement of the movable member 240, a variation in fuel pressure within the pressure control chamber 30, a displacement of the control piston 12, a rate of injection during one cycle of injection, respectively. Solid lines indicate parameters when the first orifice 66 has a smaller diameter showing a greater flow resistance, while broken lines indicate parameters when the first orifice 66 has a greater diameter showing a smaller flow resistance.

The injection characteristics of the fuel injector 1 are almost determined by the flow rate of fuel flowing into the pressure control chamber 30 from the first orifice 66 and the flow rate of fuel flowing out of the pressure control chamber 30 into the low-pressure fuel chamber 68 through the second orifice 66. Of the injection characteristics, the start time of injection and an increase in injection rate during an early part of injection are determined by a difference in flow rate between the fuel entering the pressure control chamber 30 and the fuel emerging from the pressure control chamber 30 into the low-pressure fuel chamber 68 after the solenoid valve 20 is turned on or opened. Specifically, variations in flow rate characteristic of the first and second orifices 66 and 67 will cause a dropping speed of the pressure within the pressure control chamber 30 immediately after the solenoid valve 20 is opened to be changed. Thus, if there is a variation in flow rate characteristic of either of the first and second orifices 66 and 67, it will cause a time duration from energization of the solenoid valve 20 until the fuel pressure reaches a level at which the control piston 12 is moved in the spray hole-opening direction to be changed, thus resulting in a change in start time of injection.

As shown in Fig. 7(b), the dropping speed of pressure within the pressure control chamber 30 when the first orifice 66 shows a greater flow resistance, as indicated by the solid line, is higher than that when the first orifice 66 shows a smaller flow resistance, as indicated by the broken line. Additionally, the injection beginning is earlier and the increase in injection rate during the early part of injection is greater than those when the first orifice 66 shows the smaller flow resistance.

When the fuel pressure within the pressure control chamber 30 drops and reaches a valve-opening pressure initiating the upward movement of the control piston 12, the control piston 12 is moved in the spray hole-opening direction, and then the force acting on the pressure control piston 12 in the spray hole-opening direction will be balanced statically with that in the spray hole-closing direction. The fuel pressure within the pressure control chamber 30, however, continues to drop since the flow resistance of the second orifice 67 is set smaller than that of the first orifice 66, and the flow rate of fuel flowing out of the pressure control chamber 30 is greater than that of fuel entering the pressure control chamber 30. The static balance of the fuel pressures acting on the control piston 12 is, thus, lost so that the fuel pressure acting on the control piston 12 in the spray hole-opening direction becomes greater than that in the spray hole-closing direction, which will cause the pressure control piston 12 to be lifted upward until the fuel pressures in the spray hole-opening and -closing directions are balanced with each other. This step is repeated until the amount of lift of the control piston 12 reaches a given value. The pressure within the pressure control chamber 30 is almost maintained constant during a valve-opening stroke (i.e., upward movement) of the control piston 12. This constant pressure and the valve-opening pressure acting on the control piston 12 are determined by differences between pressure-energized areas of the needle valve 220 and the control piston 12 on which the fuel pressures act in the spray hole-opening and -closing directions and the spring pressure of the coil spring 223 urging the needle valve 220 in the spray hole-closing direction, and not the flow rate characteristics of the first and second orifices 66 and 67. The duration for which the fuel pressure within the pressure control chamber 30 is maintained constant is the time required for the control piston 12 to reach a fully-lifted position and may be changed by changing the flow rate characteristics of the first and second orifices 66 and 67. Specifically, as shown in Fig. 7(b), the duration for which the fuel pressure within the pressure control chamber 30 is kept constant when the first orifice 66 shows the smaller flow resistance indicated by the broken line is longer than that when the first orifice 66 shows the greater flow resistance indicated by the solid line.

When the control piston 12 reaches the fully-lifted position, the pressure within the pressure control chamber 30 drops below the valve-opening pressure of the needle valve 220 or down to a pressure level which is determined by the difference in flow rate characteristic between the first and second orifices 66 and 67 and is kept constant. Within this constant pressure range, the rate of injection is almost kept constant as long as the pressure acting on the top portion of the needle valve 220 is at a fixed level.

When the coil 24 is turned off to close the solenoid valve 20 after a lapse of a given period of time, the pressure within the pressure control chamber 30 rises up to a valve-closing pressure which is determined, similar to the valve-opening pressure, by the differences between

pressure-energized areas of the needle valve 220 and the control piston 12 on which the fuel pressures act in the valve-opening and -closing directions and the spring pressure of the coil spring 223 urging the needle valve 220 in the valve-closing direction. When the pressure within the pressure control chamber 30 reaches the valve-closing pressure, the control piston 12 is moved in the valve-closing direction. Specifically, when the coil 24 is deenergized, the movable member 340 is moved downward, as viewed in Fig. 2, by the spring pressure of the coil spring 27. As the movable member 340 is moved in the downward direction which closes the second orifice 67, the flow rate of fuel flowing out of the second orifice 67 is decreased so that the control piston 12 is moved in the valve-closing direction before the ball 243 closes the second orifice 67 completely.

The valve-closing pressure of the control piston 12 is, similar to the valve-opening pressure, constant even if the flow rate characteristics of the first and second orifices 66 and 67 are changed. The time interval between deenergization of the solenoid valve 20 and a time when the pressure within the pressure control chamber 30 reaches the valve-closing pressure of the control piston 12 will, however, change if the pressure within the pressure control chamber 30 during the energization of the solenoid valve 20 is changed by changes in flow rate characteristic of the first and second orifices 66 and 67. Further, the time required for closing the spray hole 101a in the valve-closing stroke of the control piston 12 is changed, similar to the valve-opening stroke, by the difference in flow rate of fuels flowing through the first and second orifices 66 and 67. The time required for closing the spray hole 101a when the first orifice 66 shows the greater flow resistance, as indicated by the solid line in Fig. 7(c), is longer than that when showing the smaller flow resistance, as indicated by the broken line. In other words, a decrease in rate of injection at termination of fuel injection when the first orifice 66 shows the greater flow resistance is slower than that when showing the smaller flow resistance.

As will be apparent from the above discussion, an increase in flow resistance of the first orifice 66 without changing the flow rate characteristic of the second orifice 67 will cause the injection beginning to be advanced and the rate of initial injection to be increased, while it retards the injection end and prolongs the injection cut-off period. Conversely, a decrease in flow resistance of the first orifice 66 without changing the flow rate characteristics of the second orifice 67 will cause the injection beginning to be retarded and the rate of initial injection to be decreased, while it advances the injection end and shortens the injection cut-off period.

The injection characteristics other than the injection cut-off period depend upon the difference in flow rate of fuels flowing into the first orifice 66 and out of the second orifice 67. Therefore, a change in flow resistance of the second orifice 67 without changing the flow resistance of the first orifice 66 also causes the injection beginning, the rate of initial injection, and the injection end to be changed. The injection cut-off period is changed only by changing the flow resistance of the first orifice 66.

In the first embodiment as described above, the first and second orifice plates 210 and 211 are made of separate members, which allows the flow rate characteristics of each of the first and second orifices 66 and 67 to be adjusted in an injection characteristic adjustment process when the fuel injector 1 is assembled by replacing corresponding one of the first and second orifice plates 210 and 211. Specifically, the injection beginning, the rate of initial injection, the injection end, and the injection cut-off period may be adjusted only by replacing one of the first and second orifice plates 210 and 211.

It is necessary to determine the flow rate characteristics of spare orifice plates before replaced with the first and second orifice plates 210 and 211. In the first embodiment, the flow rate characteristics of each spare orifice plate is determined by passing a gas oil that is fuel for diesel engines through an orifice thereof at 10 Mpa to measure the flow rate of the gas oil. After assembly of the fuel injector 1, the flow rate characteristics of the first and second orifice plates 210 and 211 may be determined by monitoring variations in rate of injection, pressure within the pressure control chamber 30, and lift of the needle valve 220.

Fig. 8 shows the fuel injector 1 according to the second embodiment of the invention. The same reference numbers as employed in the first embodiment refer to the same parts, and explanation thereof in detail will be omitted here.

The first orifice plate 56 has the first orifice 76 formed in a bottom surface exposed to the pressure control chamber 30. Specifically, the first orifice 76 is, unlike the first embodiment, exposed directly to the pressure control chamber 30, but identical in operation with the first embodiment.

The movable member 80 of the solenoid valve 20 includes the valve shaft 81, the hollow rod 82, the plunger 83, the ball 243, and the support 244. An assembly of the rod 82 and the plunger 83 corresponds to the plunger 242 of the first embodiment. The connector 84 which supplies the power to the coil 24 of the solenoid valve 20 extends diagonally up to the right in the drawing because the screw 90, as will be described in detail below, is mounted along a longitudinal center line of the solenoid valve 20.

The screw 90 is inserted into the housing 92 through the gasket 91. The amount of insertion of the screw 90 may thus be adjusted by changing the thickness of the gasket 91, which allows the spring load of the coil spring 27 acting on the plunger 83 to be regulated from outside the fuel injector 1. Specifically, the second embodiment is designed to change the injection characteristics easily by adjusting the thickness of the gasket 91.

Figs. 9 and 10 show the third embodiment of the invention which is different from the above embodiments only in structure of the first and second orifice plates. Other arrangements are identical, and explanation thereof in detail will be omitted here.

The first orifice plate 100, as shown in Fig. 10, has formed therein through holes 100a and 100b. Similarly, the second orifice plate 101, as shown in Fig. 9, has formed therein through holes 101a and 101b. The through holes 100a, 100b, 101a, and 101b serve to fix angular positions of the first and second orifice plates 100 and 101 relative to the injector body 91 using knock pins.

The through holes 100a, 100b, 101a, and 101b are arranged in the first and second orifice plates 100 and 101 so as to satisfy the following two geometrical specifications:

(1) lines extending through the through holes 100a and 100b and the through holes 101a and 101b are offset from the centers of the first and second orifice plates 100 and 101, respectively
(2) if intervals between the centers of the first and second orifice plates 100 and 101 and the through holes 100a and 101a are defined as **a**, and intervals between the centers of the first and second orifice plates 100 and 101 and the through holes 100b and 101b are defined as **b**, then **a** > **b**.

These specifications make it possible to fix angular positions of the fuel passages 100c and 101c when the first and second orifice plates 100 and 101 are Incorporated within the injector body 91 during assembly so that the fuel passages 100c and 101c are aligned with each other. Specifically, if the first and second orifice plates 100 and 101 are placed within the injector body 91 incorrectly in angular position or one of the first and second orifice plates 100 and 101 is reversed, then the knock pins cannot be inserted into the through holes 100a, 100b, 101a, and 101b, which enables the operator to perceive that there is an error in assembly.

Each of the first and second orifice plates 100 and 101 may alternatively have formed therein three or more through holes and be designed to satisfy only the above second specification (2).

Figs. 11(a) and 11(b) show the fourth embodiment of the invention which is different from the above embodiments in structure of the second orifice plate 211. Other arrangements are identical, and explanation thereof in detail will be omitted here. Fig. 11(b) shows only central portions of the first and second orifice plates 210 and 211 different from those in the above embodiments for the brevity of illustration.

The second orifice plate 211 has, as shown in Fig. 11(b), a cylindrical fuel chamber 168 formed in an upper surface thereof coaxially with the second orifice 67 in communication with the second orifice 67. The cylindrical fuel chamber 168 is greater in diameter, that is, smaller in flow resistance than the second orifice 67 and establishes fluid communication between the second orifice 67 and the low-pressure fuel chamber 68 when the solenoid valve 20 is turned on to lift the ball 243 upward. The cylindrical fuel chamber 168 has an opening area smaller than an area of a flat valve head 243a of the ball 243 of the solenoid valve 20.

The second orifice plate 211 has a flat valve seat 53 and a fuel relief path 54 formed on and in the upper surface thereof. The flat valve seat 53 consists of a central annular seat 53a and four fan-shaped seats 53b which are all engageble with the flat valve head 243a in surface contact. The annular seat 53a is formed around the periphery of the cylindrical fuel chamber 168. The fan-shaped seats 53b are formed at regular intervals around the annular seat 53a.

The fuel relief path 54 includes a central annular path 54a and four radially extending paths 54b and establishes fluid communication with the low-pressure fuel chamber 68 at all times. The annular path 54a is defined between an outer periphery of the annular seat 53a and inner peripheries of the fan-shaped seats 53b and coaxially with the cylindrical fuel chamber 168 for equalizing fuel pressures acting on the flat valve head 243a of the ball 243. The radially extending paths 54b are each defined between adjacent two of the fan-shaped seats 53b and communicate with the annular path 54a at angular intervals of 90°.

Formed around the fan-shaped seats 53b is the annular path 155, as shown in Figs. 6(a) and 6(b), which communicates with the radially extending paths 54b. The annular path 155 is, as described above, provided for adding a given volume to the low-pressure fuel chamber 68 to facilitate ease of the fuel flow to the low-pressure fuel chamber 68 when the ball 243 is lifted away from the second orifice plate 211.

The fourth embodiment has the following specifications on the structural elements as shown in Figs. 11(a) and 11(b):

1. diameter **a** of the first orifice 66 = Ø0.19 mm
2. diameter **b** of the second orifice 67 = Ø0.29 mm
3. diameter **c** of the cylindrical fuel chamber 168 (i.e, an inner diameter of the annular seat 53a) = Ø0.4 mm
4. inner diameter **d** of the annular path 54a (i.e., an outer diameter of the annular seat 53a) = Ø0.7 mm
5. outer diameter **e** of the annular path 54a = Ø1.2 mm
6. depth of the annular path 54a = 0.1 mm
7. width of the paths 54b = 0.4 mm
8. depth of the paths 54b = 0.1 mm
9. diameter **f** of the ball 243 = Ø2.0 mm
10. diameter **g** of the flat valve head 243a = Ø1.63 mm
11. diameter **h** of the control piston 12 = Ø5.0 mm
12. stroke of the movable member 240 = 0.1 mm
13. diameter of the needle valve 220 = Ø4.0 mm

14. seat diameter of the needle valve 220 (i.e., the diameter of a seat area of a head of the needle valve 220 exposed to the spray hole 101a) = Ø2.25 mm

15. set load of the spring 27 = 50 N

16. set load of the spring 223 = 40 N

In operation of the fuel injector 1, when the coil 24 of the solenoid valve 20 is in an off-position, the plunger 242 is urged downward, as viewed in Fig. 2, by the spring pressure of the coil spring 27. The ball 243 is seated on the second orifice plate 211 to block the fluid communication between the pressure control chamber 30 and the low-pressure fuel chamber 68.

Even when the ball 243 is slightly separated from the second orifice plate 211 with an extremely small clearance of less than 1μm causing penetration of the fuel as well as when the ball 243 is seated on the second orifice plate 211 completely, the fuel within the fuel relief path 54 is drained to the low-pressure fuel chamber 68, and the pressure thereof is kept at a low level (i.e., a drain line pressure) since the annular path 54a is formed around the annular seat 53a and communicates with the radially extending paths 54b. The pressure distribution between contact surfaces of the flat valve head 243a of the ball 243 and the annular seat 53a is expressed by a logarithmic function showing the point symmetry in which a peak pressure that is the pressure within the pressure control chamber 30 (i.e., the pressure within the cylindrical fuel chamber 168) is developed at the inner edge of the annular seat 53a, and the lowest pressure appears at the outer edge of the annular seat 53a that is the pressure within the radially extending paths 54b. If the fuel relief path 54 is not formed in the second orifice plate 211, the pressure distribution of the logarithmic function is developed over the flat valve head 243a, so that a greater fuel pressure acts on the ball 243 in the valve-opening direction when the solenoid valve 20 is turned off to close the second orifice 67. Specifically, the fuel relief path 54 serves to keep the fuel pressure lifting the ball 243 away from the second orifice plate 211 at low level when the solenoid valve 20 is in the off-position.

In this embodiment, the inner diameter **c** of the annular seat 53a is, as described above, 0.4 mm, and the outer diameter of the annular seat 53a is 0.7 mm. When the fuel pressure supplied from the common rail 141 (i.e., the pressure within the pressure control chamber 30) is 150 Mpa, the fuel pressure urging the ball 243 in the valve-opening direction will be 35 N in view of the fuel pressure distributed between the flat valve head 243a of the ball 243 and the annular seat 53a in addition to the fuel pressure within the cylindrical fuel chamber 168. The set load of the coil spring 27 is, as described above, 50 N which is greater than the fuel pressure of 35 N urging the ball 243 in the valve-opening direction. Thus, the movable member 240 is held from being lifted upward as long as the coil 24 is deenergized.

Since the diameter of the control piston 12 is 5.0 mm, the diameter of the needle valve 220 is 4.0 mm, and the seat diameter of the needle valve 220 is 2.25 mm, a pressure-energized surface of the control piston 12 is greater than that of the needle valve 220, and a difference therebetween is approximately 11 mm^2. The spring pressure of the coil spring 223 acts on the needle valve 220 in the spray hole-closing direction. Thus, the sum of a force acting on the control piston 12 in the spray hole-closing direction, produced by the fuel pressure within the pressure control chamber 30 and the spring pressure of the coil spring 223 is kept greater than the fuel pressure within the fuel sump 324 lifting the needle valve 220 upward as long as the coil 24 is deenergized, so that the needle valve 220 closes the spray hole 101a.

When the coil 24 of the solenoid valve 20 is energized, it produces an electromagnetic force of approximately 60 N attracting the armature 26, so that the sum of the electromagnetic force and the fuel pressure within the pressure control chamber 30 urging the movable member 240 in the valve-opening direction becomes greater than the spring pressure of the coil spring 27, thereby lifting the movable member 240 upward to move the ball 243 away from the second orifice plate 211. This establishes the fluid communication between the second orifice 67 and the low-pressure fuel chamber 68 so that the fuel within the pressure control chamber 30 flows into the low-pressure fuel chamber 68 through the second orifice 67.

The diameter of the cylindrical fuel chamber 168 is, as already described, greater than that of the second orifice 67, so that the flow resistance drops as the fuel flows from the second orifice 67 to the cylindrical fuel chamber 168. Therefore, even if the amount of lift of the movable member 240 is decreased below that in the above embodiments, the flow resistance of fuel flowing out of the cylindrical fuel chamber 168 may be kept smaller than that of fuel passing through the second orifice 67.

When the fuel pressure within the pressure control chamber 30 drops, and the sum of the fuel pressure within the pressure control chamber 30 urging the control piston 12 in the spray hole-closing direction and the spring pressure of the coil spring 223 becomes smaller than the fuel pressure within the fuel sump 324 lifting up the needle valve 220, it will cause the needle valve 220 to be moved away from the spray hole 101a to initiate fuel injection.

When a given injection end is reached, the coil 24 of the solenoid valve 20 is deenergized, so that the electromagnetic force attracting the armature 26 is decreased from 60 N to zero (0). This causes the movable member 240 to be moved by the spring force of the spring 27 away from the coil 24 to bring the ball 243 into engagement with the second orifice 67, thereby causing the needle valve 220 to be moved downward to close the spray hole 101a so that the fuel injection is termi-

nated.

As will be apparent from the above discussion, the fourth embodiment features the formation of the cylindrical fuel chamber 168 downstream of the second orifice 67 which shows the flow resistance smaller than that of the second orifice 67. This allows the amount of lift of the movable member 240 to be decreased, thereby resulting in improved response rate and wear resistance and decrease in mechanical noise of the fuel injector 1. Specifically, a variation in amount of lift of the movable member 240 is minimized, thus reducing a variation in flow rate of fuel flowing into the low-pressure fuel chamber 68 when the solenoid valve 20 is turned on to open the spray hole 101a.

The fourth embodiment also features the formation of the fuel relief path 54 in the upper surface of the second orifice plate 211, which decreases the fuel pressure acting on the ball 242 of the solenoid valve 20 in the valve-opening direction when the solenoid valve 20 is turned off This allows the spring pressure of the coil spring 27 urging the movable member 240 downward to be decreased, thereby also allowing the electromagnetic attracting force produced by the coil 24 when energized to be decreased.

The annular path 54a is formed in the second orifice plate 211 coaxially with the cylindrical fuel chamber 168, thereby causing the fuel pressures acting on the flat valve head 243a of the ball 243 in the valve-opening direction to be equalized to minimize inclination of the flat valve head 243a relative to the valve seat 53 of the second orifice plate 211. This allows the injection quantity to be adjusted finely.

The cylindrical fuel chamber 168 may be first drilled to guide drilling of the second orifice 67. This facilitates easy of machining of the second orifice 67.

Figs. 12(a) and 12(b) shows the fifth embodiment of the invention which is a modification of the fourth embodiment. The same reference numbers as employed in Figs. 11(a) and 11(b) refer to the same parts.

The ball 243 has formed in the flat valve head 243a a central cylindrical fuel chamber 243b which corresponds to the cylindrical fuel chamber 168 of the fourth embodiment. The cylindrical fuel chamber 243b has the diameter **k** greater than the diameter **b** of the second orifice 67. In practice, the diameter **k** = ∅0.4 mm, and the diameter **b** = ∅0.29 mm. The other dimensions **a**, **b**, **d**, **e**, **g**, and **h** are the same as those in the fourth embodiment. An area of an opening of the cylindrical fuel chamber 243b is smaller than an area of the flat valve head 243a.

The second orifice 67 opens directly to an annular seat 53c formed on the upper surface of the second orifice plate 211 so that the inner diameter of the annular seat 53c is equal to the diameter **b** of the second orifice 67. The width of the annular seat 53c is greater than that of the annular seat 53a as shown in Fig. 11(b).

Since the diameter **k** of the cylindrical fuel chamber 243b is greater than the diameter **b** of the second orifice 67, the flow resistance of fuel flowing out of the second orifice 67 becomes smaller than when the cylindrical fuel chamber 243b is not formed in the flat valve head 243a. Specifically, the fuel flowing out of the second orifice 67, like the first embodiment, is not decreased in flow rate when passing between the annular seat 53c and the flat valve head 243a. This results in improved response rate and wear resistance and decrease in mechanical noise of the fuel injector 1.

The cylindrical fuel chambers 168 and 243b, as shown in Figs. 11(b) and 12(b), may be of cone-shape in which the inner diameter increases as approaching the opening. The cylindrical fuel chamber 168 may also be formed in the second orifice plate 211 of the fifth embodiment, while the cylindrical fuel chamber 243b may also be formed in the flat valve head 243a of the fourth embodiment.

While the present invention has been disclosed in terms of the preferred embodiment in order to facilitate a better understanding thereof, it should be appreciated that the invention can be embodied in various ways without departing from the principle of the invention. Therefore, the invention should be understood to include all possible embodiments and modification to the shown embodiments which can be embodied without departing from the principle of the invention as set forth in the appended claims.

Claims

1. An accumulator fuel injection apparatus for injecting high-pressure fuel stored within a common rail into an internal combustion engine comprising:

   a valve body having formed therein a fuel inlet passage and a spray hole, fuel inlet passage communicating with the common rail;
   a valve member disposed slidably within said valve body for selectively establishing and blocking fluid communication between the fuel inlet passage and the spray hole;
   a pressure control chamber formed within said valve body, said pressure control chamber being connected to the fuel inlet passage to introduce therein fuel pressure which acts on said valve member to block the fluid communication between the fluid inlet passage and the spray hole;
   a fuel pressure drain passage formed within said valve body, connected to said pressure control chamber for draining the fuel pressure out of said valve body;
   a solenoid valve selectively establishing and blocking fluid communication between said pressure control chamber and said fuel pressure drain passage;
   a first orifice plate having formed therein a first

orifice which provides a first flow resistance to fuel flowing from the fuel inlet passage into said pressure control chamber; and

a second orifice plate having formed therein a second orifice which provides a second flow resistance smaller than the first flow resistance to the fuel flowing out of said pressure control chamber into said fuel pressure drain passage when said solenoid valve establishes the fluid communication between said pressure control chamber and said fuel pressure drain passage, said second orifice plate being disposed on said first orifice plate so that thicknesswise directions thereof coincide with each other.

2. An accumulator fuel injection apparatus as set forth in claim 1, wherein the first orifice has a length extending in parallel to a thickness of said first orifice plate, and wherein the second orifice has a length extending in parallel to a thickness of said second orifice plate.

3. An accumulator fuel injection apparatus as set forth in claim 1 or 2, wherein the first and second orifices are formed by drilling said first and second orifice plates and reaming drilled holes.

4. An accumulator fuel injection apparatus as set forth in claim 1 or 2, wherein the first and second orifices are holes formed in an electron discharge method.

5. An accumulator fuel injection apparatus as set forth in any one of claims 1 to 4, wherein the first and second orifices are polished by forcing an abrasive solution made of a mixture of liquid and abrasive grain therethrough until the flow of the abrasive solution through the first and second orifices reaches a given flow rate.

6. An accumulator fuel injection apparatus as set forth in any one of claims 1 to 5, wherein each of said first and second orifice plates is made of a disc in which first and second through holes are formed, and further comprising two knock pins inserted into said valve body through the first and second through holes of said first and second orifice plates to fix angular positions of said first and second orifice plates relative to said valve body.

7. An accumulator fuel injection apparatus as set forth in claim 6, wherein the first and second through holes are formed at different intervals away from the center of each of said first and second office plates so that a line extending through the centers of the first and second through holes is offset from the center of each of said first and second orifice plates.

8. An accumulator fuel injection apparatus as set forth in any one of claims 1 to 7, further comprising a first large-diameter hole having a diameter greater than that of the first orifice, said first large-diameter hole being formed in said first orifice plate coaxially with the first orifice in communication with the first orifice.

9. An accumulator fuel injection apparatus as set forth in any one of claims 1 to 8, further comprising a second large-diameter hole having a diameter greater than that of the second orifice, said second large-diameter hole being formed in said second orifice plate coaxially with the first orifice in communication with the second orifice.

10. An accumulator fuel injection apparatus as set forth in any one of claims 1 to 9, wherein said first and second orifice plates are so disposed within said valve body that the first orifice plate is exposed at a first surface to said pressure control chamber and at a second surface opposite the first surface in contact with a first surface of said second orifice plate, and said second orifice plate is exposed at a second surface opposite the first surface to said fuel pressure drain passage, and further comprising a cylindrical chamber formed in the second surface of said second orifice plate in communication with the second office, said cylindrical chamber having a diameter greater than that of the second orifice.

11. An accumulator fuel injection apparatus as set forth in claim 10, wherein said solenoid valve includes a valve head which opens and closes the second orifice to establish and block the fluid communication between said pressure control chamber and said fuel pressure drain passage, and further comprising an annular valve seat on which the valve head of said solenoid valve is to be seated to block the fluid communication between said pressure control chamber and said fuel pressure drain passage, said annular valve seat being formed on the second surface of said second orifice plate around an opening of said cylindrical chamber.

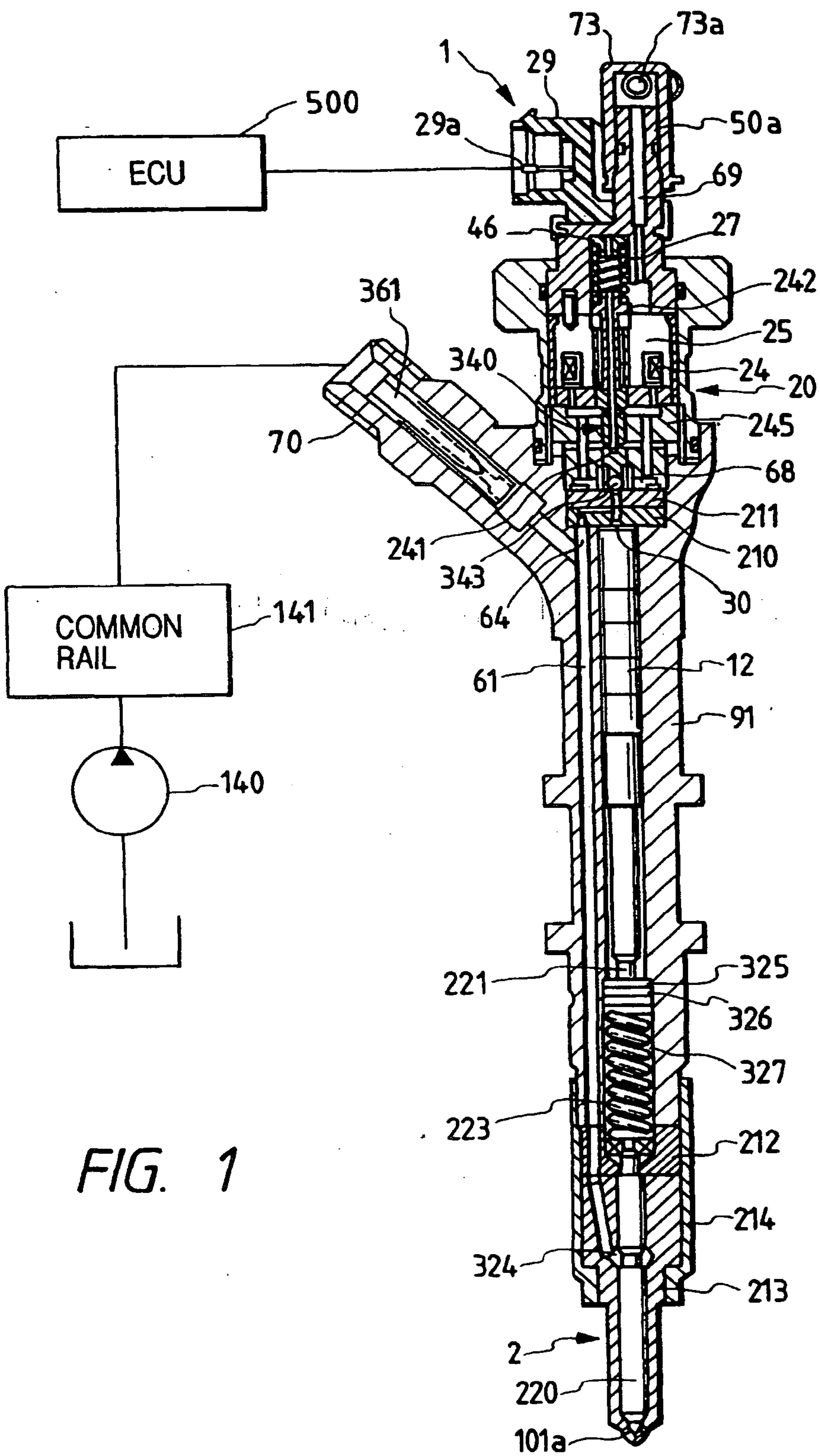
12. An accumulator fuel injection apparatus as set forth in claim 11, further comprising an annular groove formed in the second surface of said second orifice plate around said annular valve seat of said second orifice plate in fluid communication with said fuel pressure drain passage.

13. An accumulator fuel injection apparatus as set forth in any one of claims 1 to 9, wherein said solenoid valve includes a valve head which opens and closes the second orifice to establish and block the fluid communication between said pressure control

chamber and said fuel pressure drain passage, and further comprising a cylindrical chamber formed in the valve head opening to the second orifice of said second orifice plate, said cylindrical chamber having a diameter greater than that of the second orifice.

14. An accumulator fuel injection apparatus as set forth in claim 13, wherein said first and second orifice plates are so disposed within said valve body that the first orifice plate is exposed at a first surface to said pressure control chamber and at a second surface opposite the first surface in contact with a first surface of said second orifice plate, and said second orifice plate is exposed at a second surface opposite the first surface to said fuel pressure drain passage, and further comprising an annular valve seat on which the valve head of said solenoid valve is to be seated to block the fluid communication between said pressure control chamber and said fuel pressure drain passage, said annular valve seat being formed on the second surface of said second orifice plate around an opening of the second orifice.

15. An accumulator fuel injection apparatus as set forth in claim 14, further comprising an annular groove formed in the second surface of said second orifice plate around said annular valve seat of said second orifice plate in fluid communication with said fuel pressure drain passage.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24
20
70
245
68
211
241
343
210
64
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12
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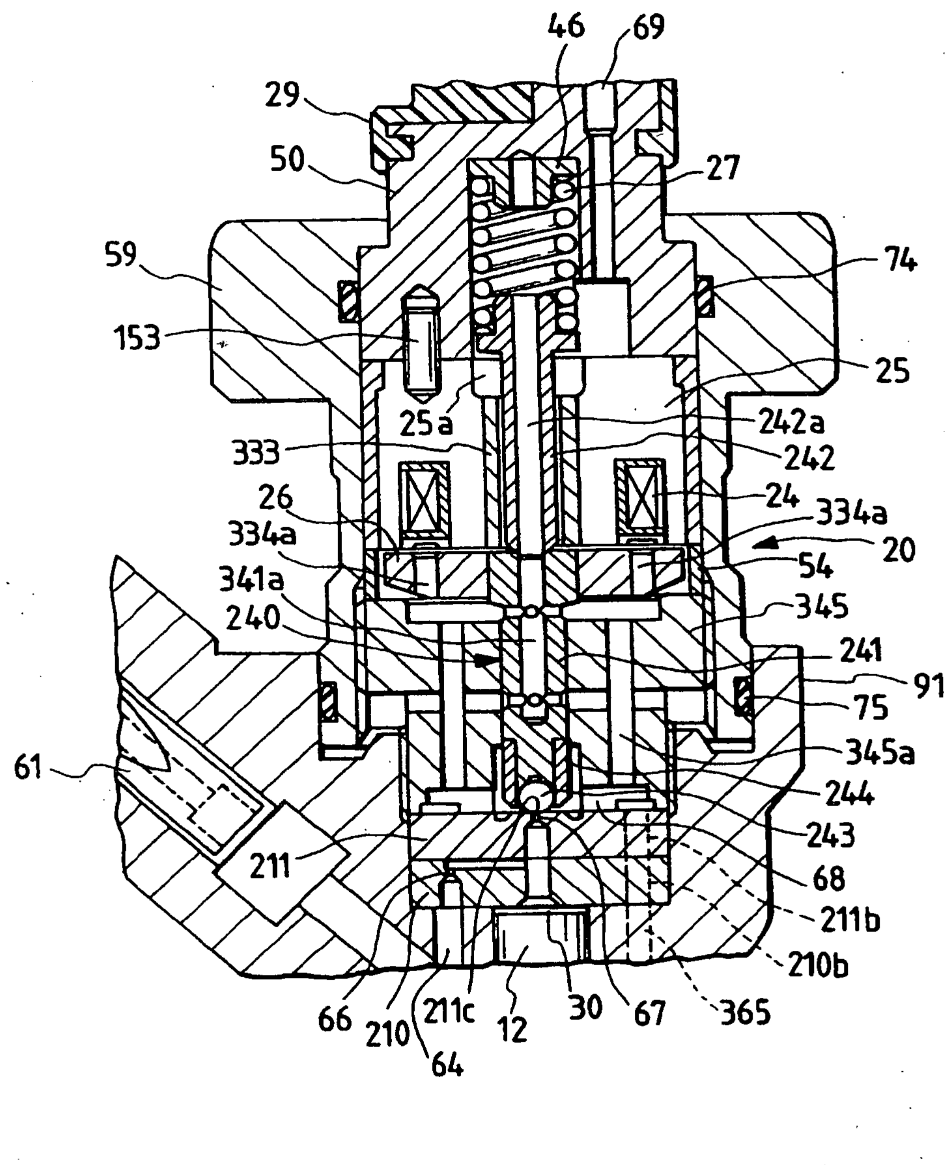
327
223
212
214
324
213
2
220
101a

FIG. 1

FIG. 2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240
345
241
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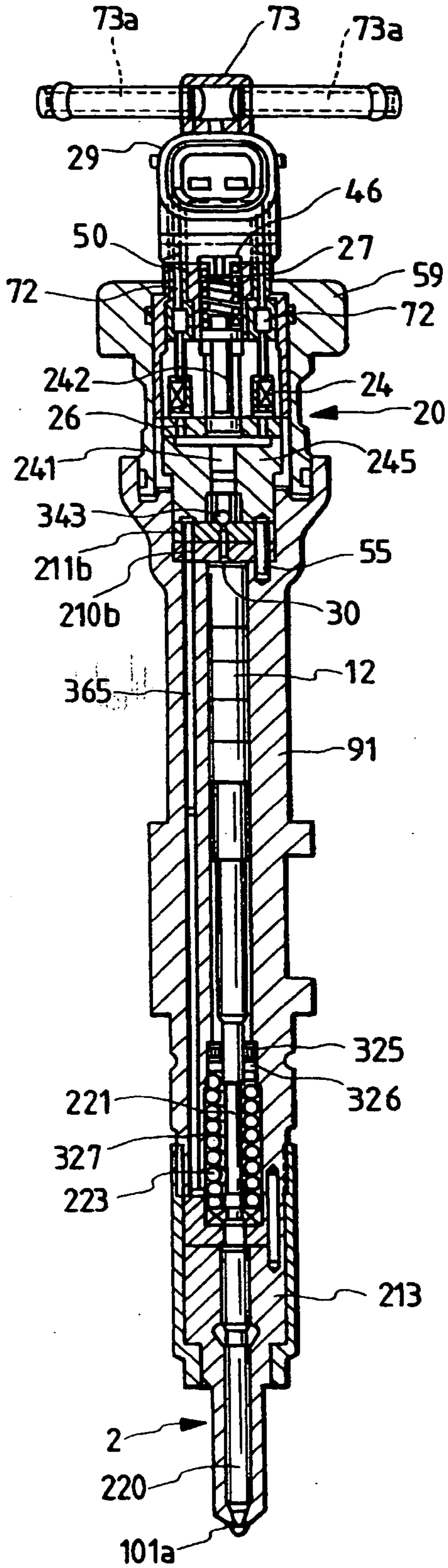
68
211b
210b
66
210
211c
12
30
67
365
64

FIG. 3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24
26
20
241
245
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210b
30
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365
91
325
221
326
327
223
213
2
220
101a

FIG. 4

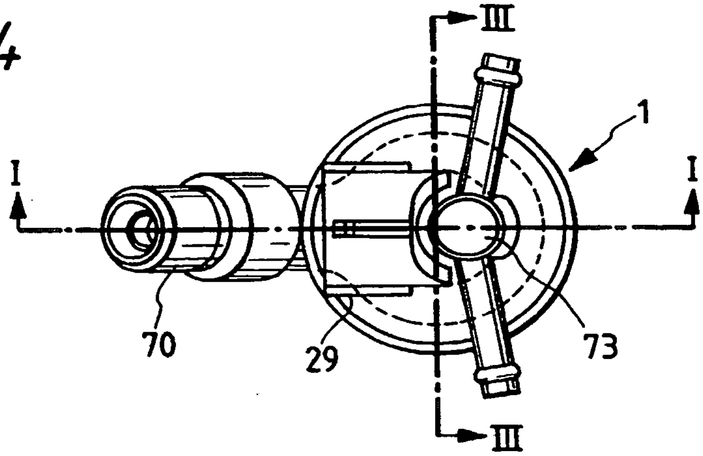


FIG. 5(a)

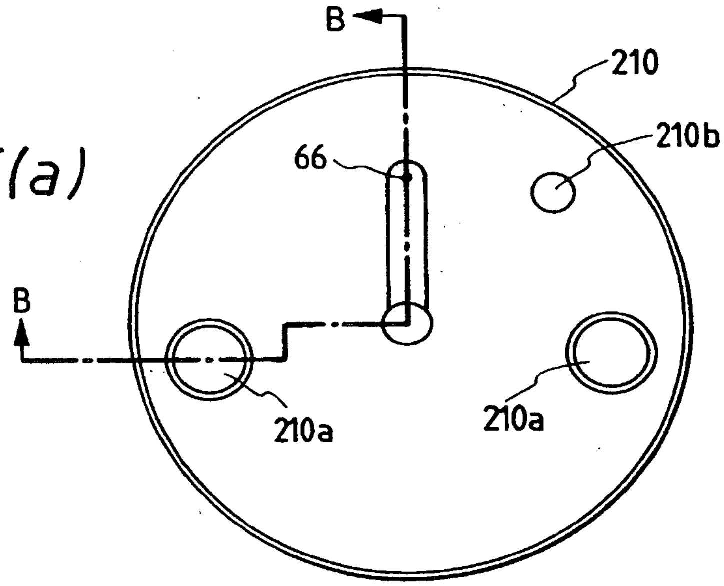


FIG. 5(b)

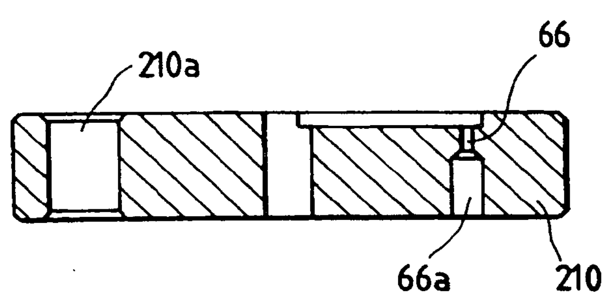


FIG. 6(a)

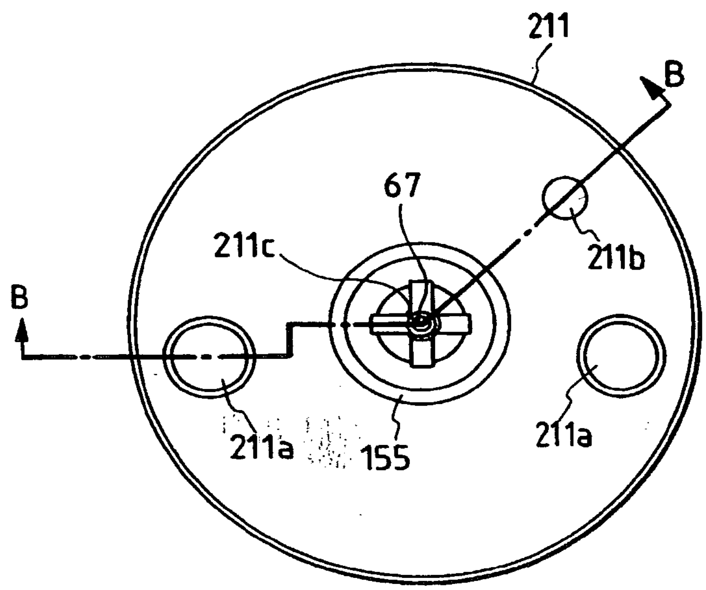

211
B
67
211c
211b
B
211a
155
211a

FIG. 6(b)

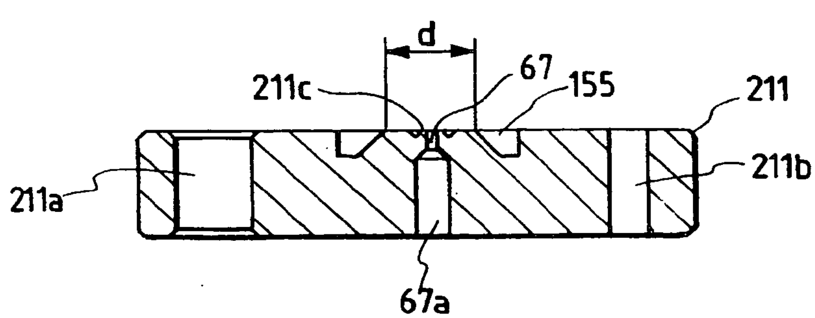

d
67
211c
155
211
211b
211a
67a

FIG. 7(a)

FIG. 7(b)

FIG. 7(c)

FIG. 7(d)

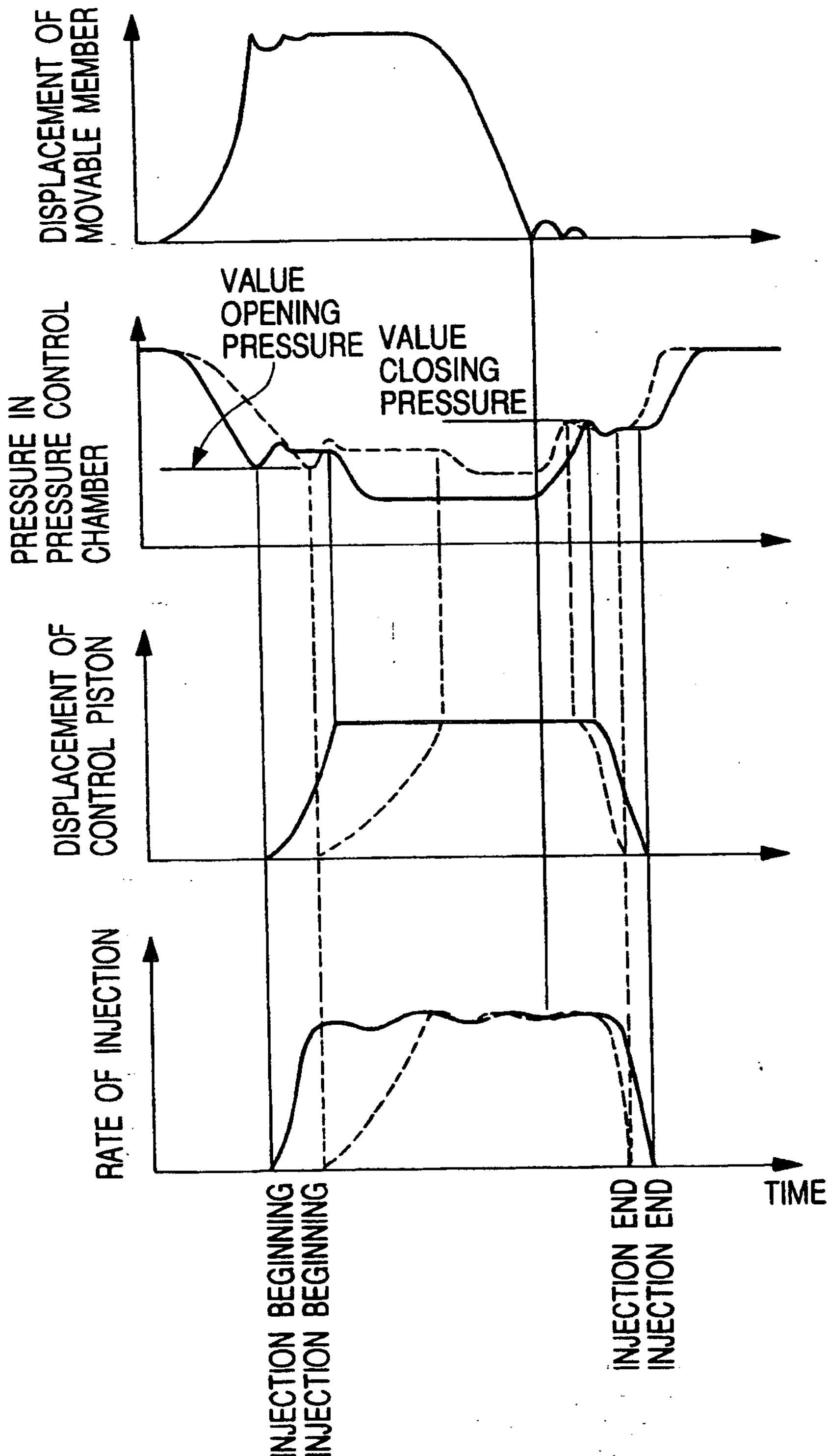
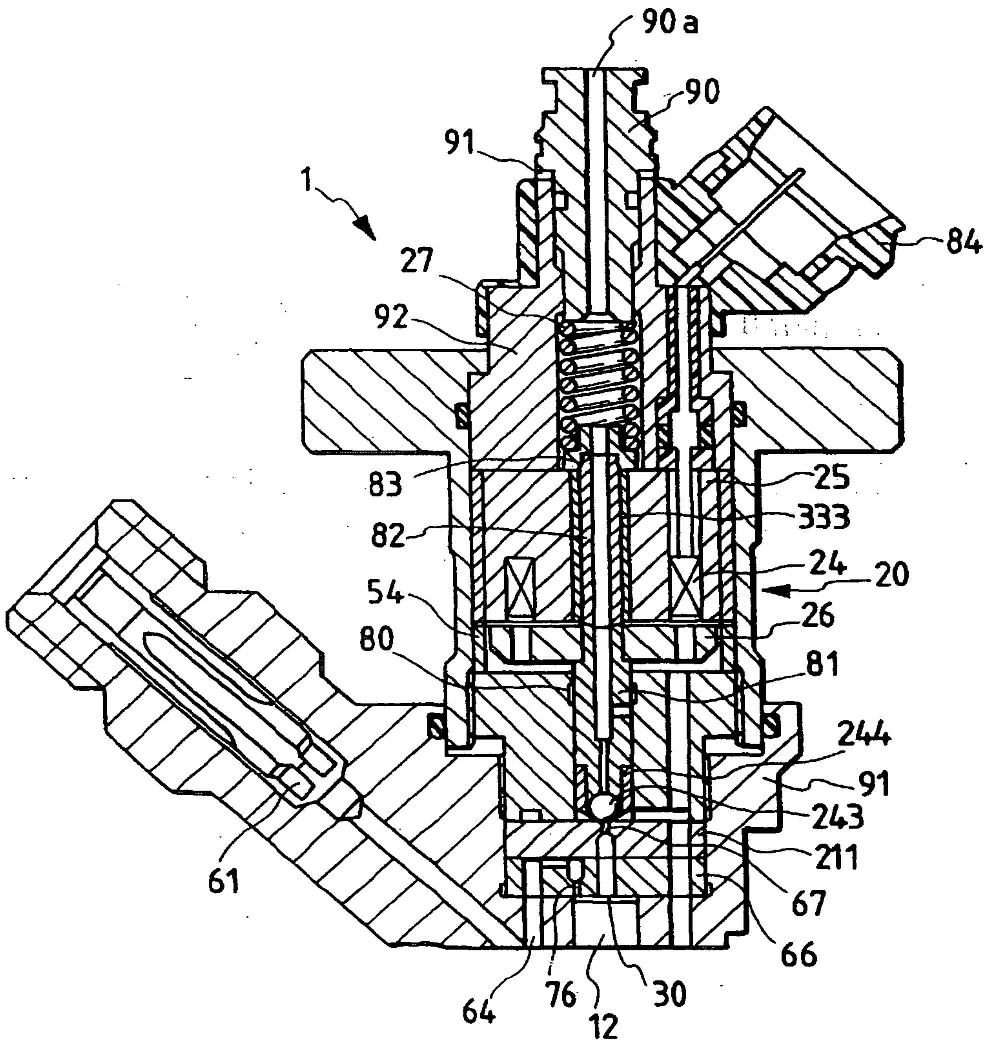


FIG. 8


90 a
90
91
1
84
27
92
83
25
333
82
24
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54
26
80
81
244
91
243
211
61
67
66
64
76
12
30

FIG. 9

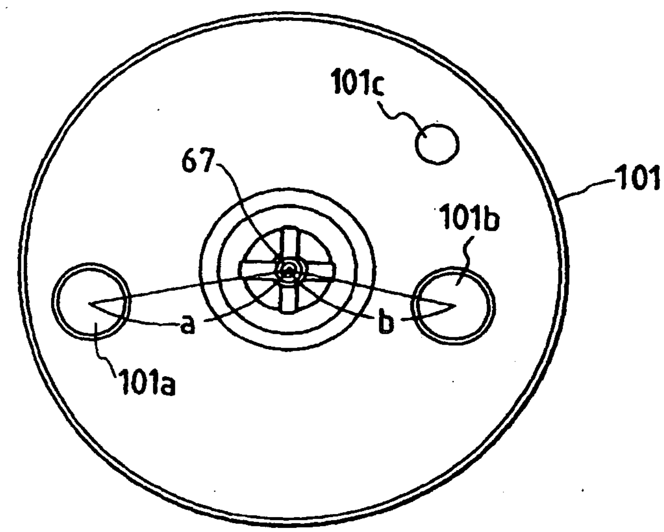


FIG. 10

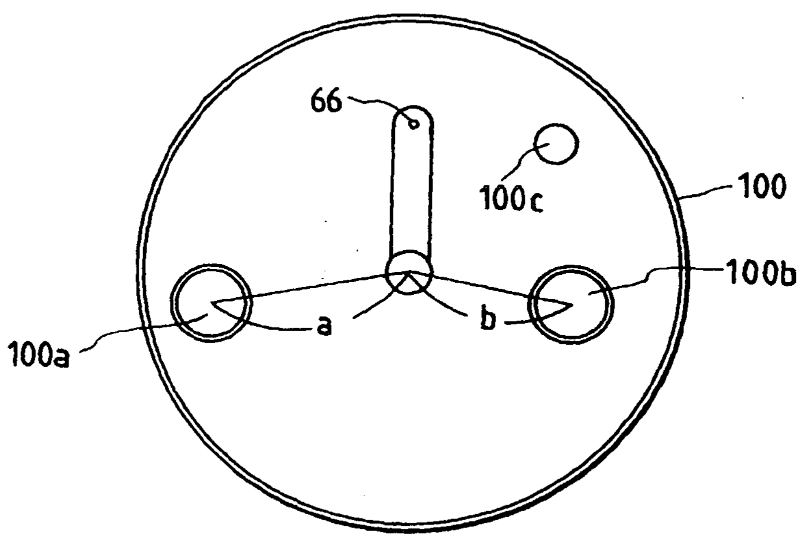


FIG. 11(a)

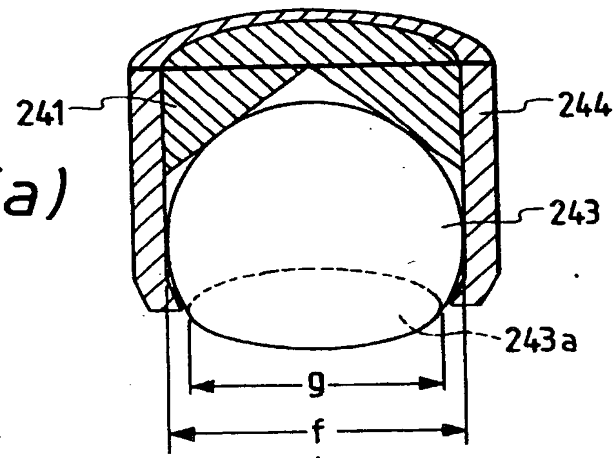


FIG. 11(b)

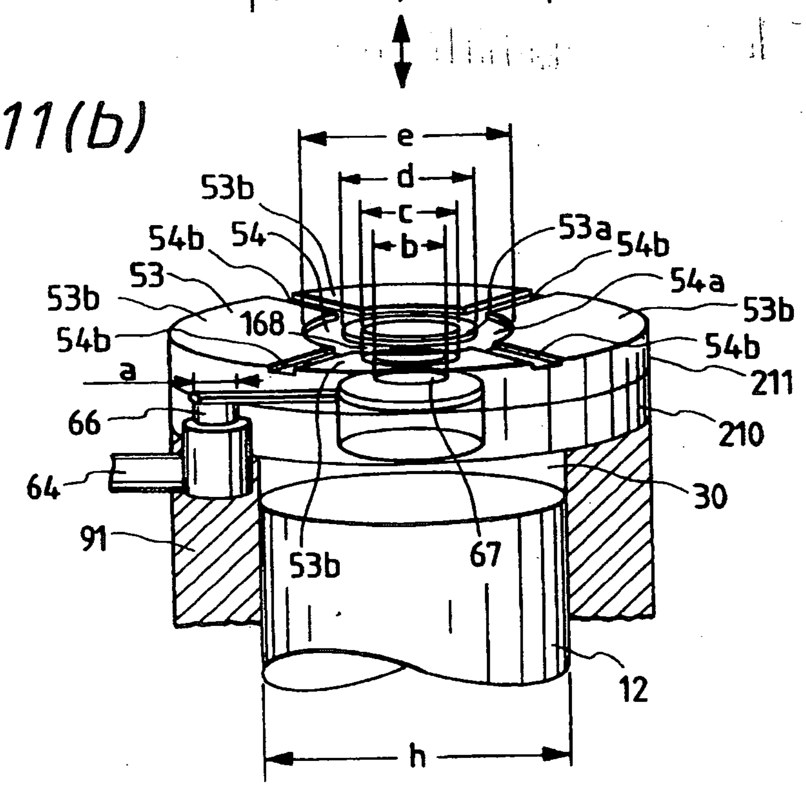


FIG. 12(a)

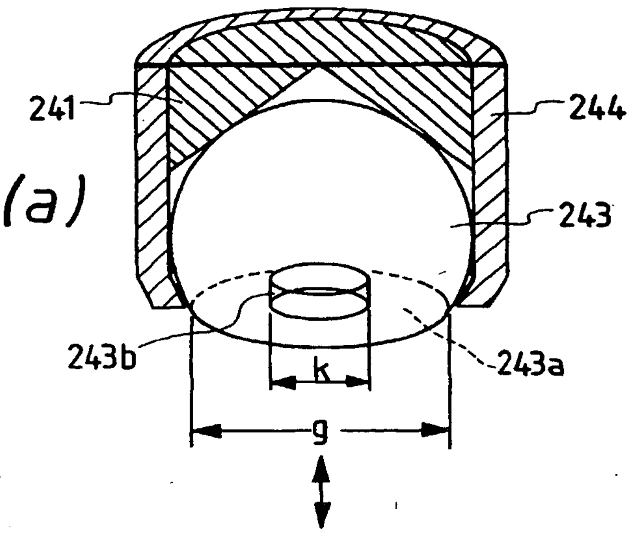
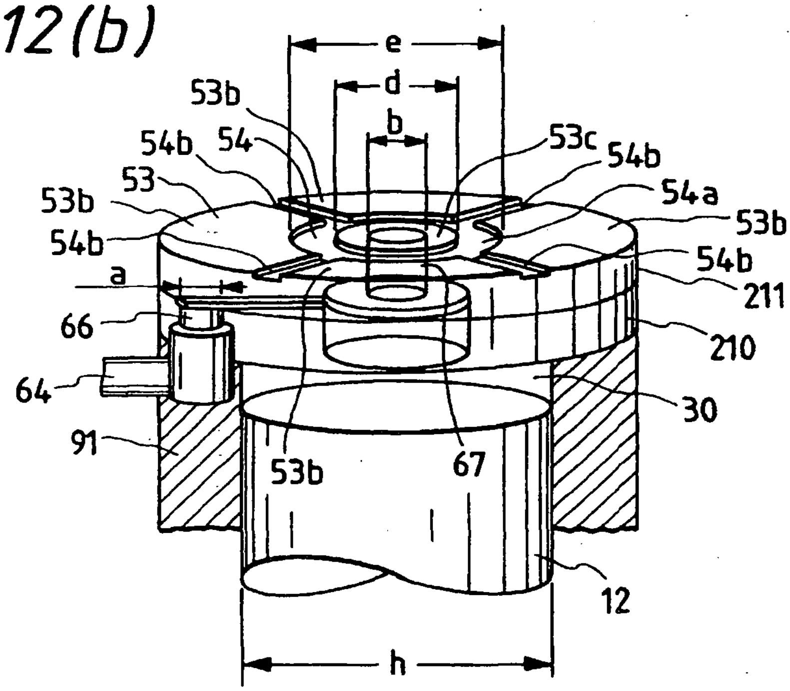


FIG. 12(b)



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EUROPEAN SEARCH REPORT

Application Number
EP 97 12 0502

DOCUMENTS CONSIDERED TO BE RELEVANT

Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int.Cl.6)
X,P	PATENT ABSTRACTS OF JAPAN vol. 097, no. 010, 31 October 1997 & JP 09 158811 A (DENSO CORP), 17 June 1997, * abstract *	1	F02M47/02
A	US 4 566 416 A (BERCHTOLD MAX) * column 3, line 30 - line 68; figure 2 *	1	
A	EP 0 740 068 A (LUCAS IND PLC) * column 2, line 29 - column 3, line 25; figure 2 *	1	
A	GB 2 185 530 A (DERECO DIESELMOTOREN FORSCHUNG;IVECO FIAT) * page 3, line 25 - line 62; figures 2,3 *	1	
A	EP 0 304 747 A (WEBER SRL) * the whole document *	1	

TECHNICAL FIELDS SEARCHED (Int.Cl.6)
F02M

The present search report has been drawn up for all claims

Place of search	Date of completion of the search	Examiner
THE HAGUE	26 February 1998	Friden, C

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